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REPORT

Some aspects of the transmission of sound through panels and of outside broadcast vehicle sound insulation

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SOME ASPECTS OF THE TRANSMISSION OF SOUND THROUGH PANELS AND OF OUTSIDE BROADCAST VEHICLE SOUND INSULATION

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Summary

The derivation of the standard relationship between transmission loss and airborne sound insulation is discussed for the case in which the transmitting panel does not occupy the complete common wall between two rooms. The dependence of this relationship on the assumptions made in its derivation and on the nature of the sound fields on the two sides of the panel is considered, particularly for the case of the sound insulation between the exterior and interior of an Outside Broadcast vehicle. A particular relationship between transmission loss and sound insulation is put forward for this case and tests to confirm this relationship experimentally are described. Sound insulation assessments on samples of vehicle wall material fabricated using a bonding process are discussed.

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SOME ASPECTS OF THE TRANSMISSION OF SOUND THROUGH PANELS AND OF OUTSIDE BROADCAST VEHICLE SOUND INSULATION

LIST OF SYMBOLS

- c — Velocity of sound in air
- D — Energy density associated with sound field:
  - D_{DS} — Direct field in source room
  - D_{RS} — Reverberant field in source room
  - D_{DR} — Direct field in receive room
  - D_{RR} — Reverberant field in receive room
  - D_{TR} — Both fields in receive room
- E — Energy associated with sound fields:
  - E_{RS} — Reverberant field in source room
  - E_{DR} — Direct field in receive room
  - E_{RR} — Reverberant field in receive room
- ΔE_{PS} — Energy removed per reflection by panel from reverberant field in source room
- ΔE_{RR} — Energy removed per reflection by walls of receive room from reverberant field
- f — Frequency
- L_p — Sound pressure level
- L_T — Transmission loss (also termed sound reduction index)
- ΔL_p — Airborne sound insulation (also termed noise reduction or sound level reduction)
- $\Delta L'_p$ — Anomalous value of ΔL_p
- l_R — Distance between panel and opposite wall of receive room
- $\bar{l}_R$ — Mean free path between reflections in receive room
- m — Surface density
- N_P — Number of reflections per second from panel
- N_S — Number of reflections per second in source room
- N_R — Number of reflections per second in receive room
- $\bar{p}$ — Average rms sound pressure:
  - $\bar{p}_S$ — In source room
  - $\bar{p}_R$ — In receive room
- $\bar{p}_{ref}$ — Reference sound pressure
- $\bar{p}_r$ — Sound pressure at distance r from point source
  - $\bar{p}_{rD}$ — Direct field component of $\bar{p}_r$
  - $\bar{p}_{rR}$ — Reverberant field component of $\bar{p}_r$
- R_S — Room constant of source room
- R_R — Room constant of receive room
- r — Distance from point source
- S — Surface area:
  - S_S — Of source room
  - S_R — Of receive room
  - S_P — Of panel
  - S_W — Of receive room wall into which panel is set
- T — Reverberation time
- t_R — Time taken for sound to traverse length of receive room
- V — Volume:
  - V_S — Of source room
  - V_R — Of receive room
- W — Power
  - W_T — Total emitted into room by a source
  - W_D — Associated with direct sound field
  - W_R — Associated with reverberant sound field
  - W_A — Absorbed at first reflection
  - W_S — Incident onto panel
  - W_P — Removed by way of panel
  - W_{TR} — Radiated into receive room by panel
  - W_{DR} — W_D component in receive room
  - W_{RR} — W_R component in receive room
  - W_X — Emitted by point source
- $\bar{\alpha}$ — Mean absorption coefficient of room surface
  - $\bar{\alpha}_S$ — For source room
  - $\bar{\alpha}_R$ — For receive room
- α_P — Absorption coefficient of panel
- θ — Angle of incidence of sound onto surface
- ρ_0 — Density of air
- τ — Transmission coefficient of panel
- $\bar{\tau}$ — Average transmission coefficient (wall and panel)
- ω — Angular frequency

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1. Introduction

The transmission loss* (L_T) of a panel is defined by the relationship

$$L_T = 10 \log_{10} \frac{1}{\tau} \qquad (1)$$

where τ (the "transmission coefficient") is the ratio of the power transmitted through the panel to the power incident upon it. If the panel is set into the common wall between two rooms (Fig. 1), then the standard formula [1b,2a] relating the transmission loss to the airborne sound insulation or difference in the average sound pressure levels (ΔL_p)** measured in the "source" and "receive" rooms is

$$L_t = \Delta L_p + 10 \log_{10}\left(\frac{1}{4} + \frac{S_P}{R_R}\right) \qquad (2)$$

where S_P is the area of the panel and R_R is the "room constant" of the receive room. If $\bar{\alpha}_R$ and S_R are respectively the mean absorption coefficient of the surfaces of the receiving room (including air absorption, where appropriate) and the total surface area of this room, then:

$$R_R = \frac{\bar{\alpha}_R S_R}{1 - \bar{\alpha}_R} \qquad (3)$$

As it stands, Equation 2 carries the implication that, if the area of the panel is reduced to zero, the sound pressure level (spl) in the receiving room will not also become zero, but will differ from the spl in the source room by a quantity $\Delta L'_p$ where

$$\Delta L'_p = L_T - 10 \log_{10}(\tfrac{1}{4}) = L_T + 6\,\text{dB}$$

This is clearly incorrect. The anomaly arises because the assumption[1c] is made, in deriving Equation 2, that the "panel" is a complete wall of the receiving room. Unfortunately, this assumption is often not emphasized. The anomaly is often avoided by ignoring the factor "$\frac{1}{4}$" in relation to the factor S_P/R_R in Equation 2:[2a,3] in fact, with the further approximation ($R_R = S_R \bar{\alpha}_R$) made by Gilford, this approach leads to a relationship described later in this Report (Equation 38) which does not in fact depend for its validity on such approximations. In this Report the relationship between transmission loss and sound

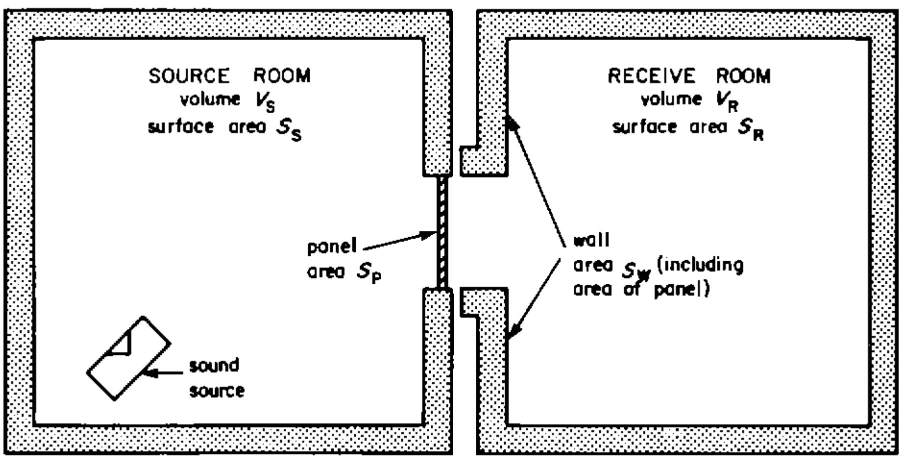

Fig. 1 – Transmission loss measuring suite.

*Sometimes termed the "sound reduction index".[2a]
**Sometimes termed the "noise reduction"[1a] or "sound level reduction".[2a]

insulation is examined without making the assumption that the panel transmitting the sound is a complete wall of the receive room.

A further assumption that is usually made when considering the transmission of sound through panels is that the sound field on the source side of the panel is "diffuse" so that sound arrives at the panel at random incidence. In an ideal transmission suite this condition can be ensured by arranging that the source room is very reverberant and the actual sound source is suitable positioned. Such conditions may not, however, always apply in practice, where the source room may have a "dead" acoustic and the sound source be positioned close to the panel under consideration. An important example of this situation is an outside broadcast vehicle, where "reverberation" plays no part in creating the source sound field. This aspect is also considered in this Report.

2. Relationship between sound pressure level difference and transmission loss for a transmitting panel of smaller area than one wall of an enclosure

2.1. Sound fields in a room

The derivation of the relationship between the difference in the average sound pressure levels existing in the source and receive rooms (Fig. 1), and the transmission loss of the panel in the common wall of the two rooms, takes into account two components of the sound field in a room:

(a) Direct Sound Field: the sound field due to emission from a source in the room, up to the time when the sound experiences its first reflection from the walls of the room. The power associated with the direct sound field (W_D) is equal to the total power emitted into the room by the source (W_T): in the following discussion the symbol W_T will be used for this quantity. The sound power (W_A) absorbed at the first reflection is given by

$$W_A = W_T \bar{\alpha} \tag{4}$$

where $\bar{\alpha}$ is the mean absorption coefficient of the room surface.

(b) Reverberant Sound Field: the total sound field due to sound reflected at least once from a surface in the receiving room. The power (W_R) associated with the reverberant sound field is that "left over" after the first reflection and is also equal to the power absorbed in all subsequent reflections. Since, after the lapse of time long enough for steady-state conditions to be established,

$$W_T = W_A + W_R \tag{5}$$

it can be seen from Equations 4 and 5 that

$$W_R = W_T(1 - \bar{\alpha}) \tag{6}$$

2.2. Receive room considerations

Consider first the direct sound field in the "receive" room, for the case in which a transmitting panel of area S_P is set into a wall of the (rectangular) room. Let the distance between the panel and the opposite wall be l_R and let the volume and total surface area of the receiving room be V_R and S_R respectively. Assume that the direct sound field fills the complete volume of the room and also that sound emerging from the panel undergoes its first reflection at the opposite wall after travelling the distance l_R (geometrical considerations show that these two assumptions are, strictly speaking, incompatible; this point is discussed later). Since, as discussed above, the total power associated with the direct sound field (W_{DR}) is equal to the power (W_{TR}) radiated by the panel, the energy (E_{DR}) associated with this field is equal to the product of this power and the time (t_R) for the sound to traverse the length of the room.

Thus:

$$E_{DR} = W_{TR} t_R \tag{7}$$

The energy density D_{DR} (energy per unit volume) associated with the direct field in the receive room is

$$D_{DR} = \frac{E_{DR}}{V_R} \tag{8}$$

Furthermore,

$$t_R = \frac{L_R}{c} \tag{9}$$

where c is the velocity of sound. From Equations 7, 8 and 9 it can be seen that

$$D_{DR} = \frac{W_{TR} L_R}{V_R c} \tag{10}$$

Turning now to the reverberant sound field in the receive room, the reverberant energy (E_{RR}) is given by

$$E_{RR} = V_R D_{RR} \tag{11}$$

where D_{RR} is the reverberant energy density. The reverberant energy removed on each reflection of the sound (ΔE_{RR}) is

$$\Delta E_{RR} = E_{RR}\bar{\alpha}_R \qquad (12)$$

The mean free path between reflections ($\bar{l}_R$) is [1d, 4]

$$\bar{l}_R = \frac{4V_R}{S_R} \qquad (13)$$

Since the distance travelled by the sound in one second is c, the number of reflections per second (N_R) is

$$N_R = \frac{c}{\bar{l}_R} \qquad (14)$$

Hence the reverberant energy removed per second, or in other words the reverberant sound power W_{RR} is

$$W_{RR} = \Delta E_{RR} N_R \qquad (15)$$

Substituting from Equations 11, 12, 13 and 14 gives

$$W_{RR} = \frac{D_{RR}\bar{\alpha}_R c S_R}{4} \qquad (16)$$

Since the receive room is under consideration, Equation 6 may be written

$$W_{RR} = W_{TR}(1 - \bar{\alpha}_R)$$

Hence substituting for W_{RR} in Equation 16 and rearranging gives

$$D_{RR} = \frac{4W_{TR}(1 - \bar{\alpha}_R)}{cS_R\bar{\alpha}_R} \qquad (17)$$

In terms of the room constant R_R (Equation 3), Equation 17 becomes

$$D_{RR} = \frac{4W_{TR}}{cR_R} \qquad (18)$$

The total energy density (D_{TR}) associated with the sound field in the receiving room is the sum of the direct and reverberant energy densities. Thus,

$$D_{TR} = D_{DR} + D_{RR} = \frac{W_{TR}}{c}\left(\frac{l_R}{V_R} + \frac{4}{R_R}\right) \qquad (19)$$

Now it can be shown[1e] that, in general, the average energy density D is related to the average sound pressure $\bar{p}$ by the expression

$$D = \frac{\bar{p}^2}{\rho_0 c^2} \qquad (20)$$

where ρ_0 is the density of air.

Hence, substituting for D_{TR} in Equation 19 (and adding the appropriate subscripts)

$$\bar{p}_R{}^2 = \rho_0 c W_{TR}\left(\frac{l_R}{V_R} + \frac{4}{R_R}\right) \qquad (21)$$

In Equation 21 $\bar{p}_R$ is strictly speaking the average r.m.s. sound pressure taken over one sound wavelength. In practical terms, the r.m.s. sound pressure is usually measured at a number of randomly-chosen positions in the room under consideration and the average value taken.

2.3. Source room considerations

It is assumed that on the source side, the panel is exposed only to the reverberant field. From Equations 13 and 14, with appropriate change of subscript to denote a reference to the source room, it can be seen that the number (N_S) of reflections per second on all walls is given by

$$N_S = \frac{cS_S}{4V_S} \qquad (22)$$

where V_S and S_S are the volume and total surface area of the source room. Assuming an equal probability of a reflection occurring on any part of any wall, the number of reflections per second (N_P) on the panel of area S_P is therefore

$$N_P = N_S \cdot \frac{S_P}{S_S}$$

$$= \frac{cS_P}{4V_S} \qquad (23)$$

The total reverberant energy (E_{RS}) in the source room, is by analogy with Equation 11,

$$E_{RS} = D_{RS} V_S \qquad (24)$$

Thus the reverberant energy removed per reflection by way of the panel (ΔE_{PS}) is

$$\Delta E_{PS} = E_{RS}\alpha_P$$

$$= D_{RS} V_S \alpha_P \qquad (25)$$

where α_P is the absorption coefficient of the panel.

Hence the reverberant energy removed per second by way of the panel, or in other words the reverberant power (W_P) removed by way of the panel, becomes

$$W_P = \Delta E_{PS} \cdot N_P$$

$$= \frac{D_{RS} c S_P \alpha_P}{4} \qquad (26)$$

from Equations 23 and 25. By substituting Equation 20 into Equation 26 it can be seen that

$$W_P = \frac{\bar{p}_S{}^2 S_P \alpha_P}{4 \rho_0 c} \qquad (27)$$

where $\bar{p}_S$ is the average sound pressure in the source room. Since the assumption has been made that the panel is exposed only to the reverberant field, measurements of $\bar{p}_S$ should be made such that the effect of the direct sound field in the source room is negligible.

In conventional terms, the absorption coefficient of the panel is a measure of the fraction of the power incident onto the panel which is absorbed by it, irrespective of the method by which this absorption takes place. In fact, some of the power is dissipated in the panel itself, while a further fraction is transmitted through the panel and emerges into the receive room. It is however convenient to regard α_P as being a measure only of this latter component; that is to say, α_P is the fraction of the power incident onto the panel which is transmitted by it into the receive room. The total absorption of the panel, including its "lossy" component, is then greater than the value that α_P indicates; this total absorption could, if necessary, be taken account of when calculating the mean absorption coefficient of the source room, but inspection of Equations 22 to 27 shows that this factor does not enter into the argument and its consideration is therefore not necessary. Using this interpretation of α_P, it can be seen that it may be directly equated with the transmission coefficient τ (see Equation 1) which is defined as the ratio of transmitted to incident power. Furthermore, the power abstracted from the sound field in the source room (W_P in Equation 27) is in these terms equal to the power radiated into the receive room (W_{TR} in Equation 21). Equation 27 can therefore be written

$$W_{TR} = \frac{\bar{p}_S{}^2 S_P \tau}{4 \rho_0 c} \qquad (28)$$

2.4. Transmission loss of the panel

From Equations 21 and 28 it can be seen that

$$\bar{p}_R{}^2 = \frac{\bar{p}_S{}^2 S_P \tau}{4} \left(\frac{l_R}{V_R} + \frac{4}{R_R} \right)$$

$$= \bar{p}_S{}^2 S_P \tau \left(\frac{l_R}{4V_R} + \frac{1}{R_R} \right) \qquad (29)$$

Equation 29 can be rearranged to give

$$\frac{1}{\tau} = \frac{\bar{p}_S{}^2}{\bar{p}_R{}^2} S_P \left(\frac{l_R}{4V_R} + \frac{1}{R_R} \right) \qquad (30)$$

or, expressed in decibel units

$$10 \log_{10} \frac{1}{\tau} = 20 \log_{10} \bar{p}_S - 20 \log_{10} \bar{p}_R$$

$$+ 10 \log_{10} S_P \left(\frac{l_R}{4V_R} + \frac{1}{R_R} \right) \qquad (31)$$

Now in general terms sound pressure level (L_p) is related to sound pressure p by the relationship

$$L_p = 20 \log_{10} \frac{\bar{p}}{\bar{p}_{ref}} \qquad (32)$$

where $\bar{p}_{ref}$ is a reference sound pressure value (20 μNm^{-2} r.m.s.). Furthermore, the term on the left-hand side of Equation 31 is the transmission loss L_T (see Equation 1). Thus Equation 31 becomes

$$L_T = \Delta L_p + 10 \log_{10} S_P \left(\frac{l_R}{4V_R} + \frac{1}{R_R} \right) \qquad (33)$$

where ΔL_p is the difference in the average sound pressure levels measured in the source and receive rooms.

Equation 33 may be used instead of the standard formula (Equation 2) to relate transmission loss with sound pressure level difference, in the case where the panel through which the sound is transmitted between the source and receive rooms does not occupy the complete area of the receive room wall into which it is set.

3. Discussion

3.1. Comparison with standard formula

Equation 33 may be written

$$L_T = \Delta L_p + 10 \log_{10} \frac{S_P}{S_W} \left(\frac{S_W l_R}{4V_R} + \frac{S_W}{R_R} \right) \qquad (34)$$

where S_W is the area of the wall of the receive room into which the panel is set. Since

$$S_W l_R = V_R$$

Equation 34 reduces to

$$L_T = \Delta L_p + 10 \log_{10} \frac{S_P}{S_W}\left(\frac{1}{4} + \frac{S_W}{R_R}\right)$$

$$= \Delta L_p + 10 \log_{10}\left(\frac{1}{4} + \frac{S_W}{R_R}\right) + 10 \log_{10} \frac{S_P}{S_W} \quad (35)$$

If the panel occupies the whole wall area, $S_W = S_P$ and Equation 35 reduces to the standard relationship (Equation 2), which it will be remembered was derived on this assumption. If $S_P < S_W$ the third term on the right-hand side of Equation 35 becomes negative, showing as expected (by changing the subject of the equation to ΔL_p) that a greater sound-pressure level difference will be obtained as the area of the panel is reduced, all other parameters remaining unchanged. It may be noted that Equation 35 may be derived by starting with the standard formula and regarding the relevant complete wall of the receive room as having an average transmission coefficient $\bar{\tau}$, where

$$\bar{\tau} = \frac{S_P}{S_W} \cdot \tau \quad (36)$$

3.2. The direct sound field in the receive room

The assumptions made in Section 2.2 that the direct sound field fills the complete volume of the room, and that all the sound emerging from the panel undergoes its first reflection at the opposite wall of the receive room, are commonly used[1c] in the derivation of the formula shown in Equation 2. In these terms (i.e. when the panel occupies a complete wall of the receive room) the first assumption is justified, but the second assumption is still rather suspect since it is unlikely that sound will emerge from the wall as a collimated beam. Diffraction effects and the possibility of many vibrational modes in the panel make it more likely that, at least over a band of frequencies such as is usually considered in sound insulation applications, the direction of emergence of the sound may be taken as random. Under such conditions the direct sound field would still fill the complete volume of the room even if the panel did not occupy the whole area of one wall, but first reflections could occur on any surface of the receive room (except on the wall containing the panel) and would not be confined to the wall opposite the panel. Under these conditions the room dimension L_R (the distance between the panel and the opposite wall) becomes irrelevant, and the distance travelled by the sound before undergoing its first reflection would depend on the particular path travelled. A better measure of the average distance travelled would be the mean free path ($\bar{l}_R$) of the receive room, as given by Equation 13. Inserting this value into Equation 33 gives

$$L_T = \Delta L_p + 10 \log_{10} S_P\left(\frac{\bar{l}_R}{4V_R} + \frac{1}{R_R}\right)$$

$$= \Delta L_p + 10 \log_{10} S_P\left(\frac{1}{S_R} + \frac{1}{R_R}\right) \quad (37)$$

By substituting for the value of the room constant R_R, as given by Equation 3, Equation 37 reduces to

$$L_T = \Delta L_p + 10 \log_{10} \frac{S_P}{\bar{\alpha}_R S_R} \quad (38)$$

For comparison with Equation 35 this may be written as

$$L_T = \Delta L_p + 10 \log_{10} \frac{1}{\bar{\alpha}_R} + 10 \log_{10} \frac{S_P}{S_R} \quad (39)$$

It can be seen that the term containing the correction for the area of the panel now relates this area to the total surface area of the room rather than to the area of the particular wall into which it is set. Intuitively this seems a more reasonable relationship since it is more plausible for the sound pressure level in a room to depend on the area of the panel transmitting the sound, irrespective of the area of the wall into which the panel is set, than for it apparently to become more effective in transmitting sound if the area of the wall into which it is set is reduced. Equations 38 and 39 also have the advantage that the correction term for the sound absorption in the receive room is simplified and the "room constant" is not invoked. This relationship is, in fact, advocated by some workers[2a,5,6] for relating transmission loss to differences in sound pressure level, and has mistakenly sometimes been taken only as an approximation to the standard relationship for cases in which the energy in the reverberant field is much greater than that in the direct field, and the absorption coefficient is small compared with unity.

Since the difference between the relationships shown in Equations 33 to 35 and Equations 38 to 39 is due to a change in the formulation of the direct sound field component, the effect of this change will be most noticeable when only this component is present. Under such conditions, the mean absorption coefficient becomes unity and the room constant infinite. Taking the case in which a complete wall of the receive room is transmitting sound (so

that the standard formula is directly applicable), Equations 2 or 35 become

$$L_T = \Delta L_p + 10 \log_{10} \tfrac{1}{4}$$

$$= \Delta L_p - 6 \qquad (40)$$

while Equation 38 becomes

$$L_T = \Delta L_p + 10 \log_{10} \frac{S_P}{S_R} \qquad (41)$$

Equations 40 and 41 show that in the case of a highly absorbent receive room the relationship between transmission loss and sound pressure level difference depends on the room geometry, if the direct sound field in the receive room is considered in terms of the mean free path of this room, rather than having a fixed relationship independent of room geometry as predicted by the standard formula. In the case of a cubic room the ratio S_P/S_R takes the value 1/6 and thus the "log" term in Equation 41 has the value -8 (to the nearest whole number). For conventionally-shaped rooms, in which the height is less than the length or width, the value of S_P/S_R for any wall will be less than 1/6 and the log term in Equation 41 correspondingly more negative. Only in cases in which the transmitting surface is bounded by the two longest dimensions of the room can the value of S_P/S_R attain (or exceed) the value $\frac{1}{4}$. In such cases the log term in Equation 41 can become equal to -6 (or perhaps even less negative) and thus comparable with the value obtained using the standard formula. In rooms of "conventional" shape this condition could sometimes apply to the floor or ceiling, but not to the walls, unless more than one wall is involved in sound transmission.

3.3. The sound field on the source side of the panel

Equations 33 and 38, as well as the standard relationship (Equation 2), assume that the panel is excited by a purely reverberant field. This implies random incidence of sound on to the panel. In some instances this assumption is clearly incorrect; in the case of sound transmission through the structure of an Outside Broadcast (OB) vehicle, for instance (see Section 3.4), the incident sound usually emanates from a nearby source, only a small diffuse component due to reflection from neighbouring objects being present. Assuming that the sound source is sufficiently far away from the panel for the wavefronts to be considered plane, and that the direction of sound arrival is normal to the panel through which sound transmission is taking place, then the total energy falling onto the panel in unit time (in other words, the sound power W_S) is equal to the energy contained in the rectangular volume in front of the panel having a cross-sectional area equal to that of the panel and a length equal to the distance travelled by sound in unit time. Thus

$$W_S = D_{DS} S_P c \qquad (42)$$

where D_{DS} is the direct energy density of the source sound field and other quantities are as defined previously. The power removed by transmission through the panel (W_P) is therefore

$$W_P = W_S \cdot \alpha_P = D_{DS} S_P c \alpha_P \qquad (43)$$

Using equation 20 it can then be seen that

$$W_P = \frac{\bar{p}_S{}^2 \alpha_P S_P}{\rho_0 c} \qquad (44)$$

where $\bar{p}_S$ is the average "free-field" sound pressure in front of the panel (in other words, measured at a sufficient distance from the panel to avoid "pressure-doubling" effects due to standing waves in front of the panel). Comparing this result with that obtained assuming a totally reverberant source field (Equation 27) it can be seen that for a given average sound pressure measured on the source side of the panel, the power incident onto the panel is greater when the source sound field is entirely direct than when it is entirely reverberant by a factor of four. This result agrees with the inference which can be deduced from the kinetic theory of gases, where it can be shown[7] that the number of collisions of molecules per unit time onto a surface, assuming normal incidence, is four times the collision rate assuming random incidence. The value of W_P given by Equation 44 can now be used instead of the value given by Equation 27 in deriving a version of Equation 28 for the case of a purely direct sound field on the source side of the panel, as discussed at the start of this section. New relationships between the transmission loss of the panel and the difference in sound pressure level on each side of it can then in turn be derived, following the arguments leading to Equations 33 and 38. A difficulty which occurs at this stage is that the absorption coefficient of a surface can increase in value as the angle of incidence is increased from zero, reaching a maximum at a particular value of this angle, since the acoustic impedance of the surface provides a better match to the "free air" impedance at some angles of incidence as compared to others. This effect gives rise to a change of absorption coefficient with change in the nature of the incident sound field (e.g. from reverberant to direct). It can be shown[8] that, for the hard surfaces that are normally involved in wall construction, the normal-incidence or direct-field absorption coefficient is less than the random-incidence or rever-

berant-field coefficient. If the surface absorption is small these coefficients differ by a factor of two. However, there is some doubt as to whether the more "efficient" collection of power by the surface, caused by the better match to the free-air impedance at some angles of incidence, will result in greater power being transmitted through the panel, or whether, on the other hand, it will simply cause more power to be dissipated in the panel, with the amount transmitted through the panel remaining the same. In other words, it is not clear whether the value of α_P in Equations 25–27 (which is equated with the transmission coefficient τ in Equation 28) is or is not subject to this increase by a factor of two. Discounting this factor for this reason, so as to adopt a "worst case" approach, the relationship (comparable with Equation 33) in which the commonly-adopted assumptions of the nature of the direct sound field are taken becomes

$$L_T = \Delta L_p + 10\log_{10} S_P\left(\frac{l_R}{4V_R} + \frac{1}{R_R}\right) + 10\log_{10} 4$$

$$= \Delta L_p + 10\log_{10} S_P\left(\frac{l_R}{4V_R} + \frac{1}{R_R}\right) + 6 \qquad (45)$$

Similarly, the relationship (comparable with Equation 38) in which the direct sound field in the receive room is derived in terms of the mean free path of this room becomes

$$L_T = \Delta L_p + 10\log_{10}\frac{S_P}{\bar{\alpha}_R S_R} + 6 \qquad (46)$$

Comparison of Equations 33 and 45 (or 38 and 46) shows that the sound pressure level difference between the two sides of a partition will be 6 dB less when a direct sound field exists on the source side of the partition (or, in other words, the sound insulation between the two sides of the partition will be 6 dB worse) as compared with the case when the source sound field is entirely reverberant. The decrease in panel absorption coefficient caused by the change from a reverberant sound field to a direct sound field on the source side of the panel may, as previously discussed, offset this reduction in sound insulation, giving in the limit a reduction in insulation of only 3 dB, and not 6 dB as implied by Equations 45 and 46. Because of the uncertainty as to whether this factor is or is not applicable in the present case, however, it seems reasonable to adopt the "worst case" approach and assume the correctness of these Equations.

The dependence of the transmission of sound by a panel on the nature of the sound field on the source side, discussed above, serves to emphasize the necessity of ensuring a well-defined type of source sound field (either reverberant or direct) when making transmission loss measurement, in order to avoid an experimental error which may be of up to 6 dB. It is by no means certain that the present BBC Research Department facility for measuring transmission loss is not subject to an error of this type (see Appendix). It is particularly unfortunate that the condition which gives rise to a reduction in sound insulation (the predominance of a direct sound field on the source side of the panel) corresponds to the case existing for OB vehicles, since sound insulation is already at a premium in such cases because of the necessity of using as lightweight a construction as possible.

If the angle of incidence of the sound (still consisting of plane wavefronts) on to the surface is θ, then Equation 42 becomes

$$W_S = D_{DS} S_P c \cos\theta \qquad (47)$$

which leads to the result analogous to Equation 45 that

$$L_T = \Delta L_p + 10\log_{10} S_P\left(\frac{l_R}{4V_R} + \frac{1}{R_R}\right) + 10\log_{10}\cos\theta + 6 \qquad (48)$$

or the result analogous to Equation 46 that

$$L_T = \Delta L_p + 10\log_{10}\frac{S_P}{\bar{\alpha}_R S_R} + 10\log_{10}\cos\theta + 6 \qquad (49)$$

Thus, for the same average source sound pressure measured on the source side of the panel, the sound pressure on the receive side of the panel diminishes as the angle of incidence increases (assuming other conditions remain unchanged) and (notionally) becomes zero when this angle equals 90° (i.e. glancing incidence). This is of course an idealised result as the excitation of travelling waves in the panel (the coincidence effect) will enhance the transmission through the panel at certain angles of incidence, as could the enhancement of panel absorption (if accompanied by a corresponding increase in transmission) at non-zero angles of incidence, discussed previously.

3.4. Sound insulation in outside broadcast vehicles

In the simplest terms an outside broadcast vehicle can be considered as a receive room in which one complete wall transmits sound. It thus satisfies the geometrical assumption under which the stan-

dard relationship (Equation 2) was derived. Furthermore, as this room has a short reverberation time, the further assumption that the reverberant sound field component may be neglected also seems plausible; thus using the standard approach Equation 40 is applicable and

$$L_T = L_p - 6$$

However, in this approach, L_T is derived assuming a reverberant field on the source side, whereas in the case of an OB vehicle the source field is predominantly direct (see Section 3.3). Equation 40 must thus be modified as discussed in Section 3.3, and this leads to the result

$$L_T = \Delta L_p - 6 + 6$$

$$= \Delta L_p \qquad (50)$$

Thus the sound pressure level difference measured between the exterior and interior of an outside broadcast vehicle will on this basis be numerically equal to the transmission loss of the vehicle structure, assuming that in the measurement of transmission loss, proper account has been taken of the type of sound field on the source side of the panel (again see Section 3.3).

Although OB vehicles have a design reverberation time of 0.3 seconds, this value is rarely achieved in practice, particularly in the mid- and high-frequency ranges, and a more typically average value would be 0.2 seconds. For an enclosure 3 m × 2.3 m × 2 m, which may be regarded as representative, the mean absorption coefficient then becomes 0.27 and the room constant takes the value 13.2. The "log" term in Equation 2 then takes the value −1.5 (instead of −6 as in Equation 40); in these terms the sound pressure level difference (or sound insulation) between the exterior and interior of an OB vehicle should be 4.5 dB less than the transmission loss. Furthermore, possibly because of significant reflections from nearby objects, sound may be incident on to surfaces of the vehicle other than the one facing the sound source, thus effectively increasing the total area of the transmitting "panel" and reducing the insulation still further. On the other hand, this same presence of significant sound reflections implies that the source field is to some extent diffuse, and this would tend to give rise to an increase in the sound insulation actually achieved; the presence of a significant direct component in the source field during the actual measurement of transmission loss using a specimen panel of vehicle wall (see Section 3.3) would further tend to restore the relationship, shown in Equation 51, that sound pressure difference and transmission loss are numerically equal. In addition, it may be argued that the relationships of the form shown in Equations 38 and 39 should be used, rather than those of Equations 2 and 33 to 35, this would also tend towards restoring the relationship shown in Equation 50. It therefore seems reasonable to accept Equation 50 as representing the relationship between sound pressure level difference and transmission loss in the case of OB vehicle work, while acknowledging that this relationship is only approximate to within, say, ±6 dB.

4. Comparisons between measured and predicted values of sound insulation

One present-day form of OB vehicle wall construction uses a "triple-skin" arrangement (Fig. 2) in which two sheets of aluminium alloy are fixed to a framework of vertical "top-hat" alloy sections and horizontal wooden members. Further vertical members are attached to one face of this composite panel, and these carry the third (inner) skin of Formica-covered hardboard. The spaces between the members are filled with preformed slabs of expanded polystyrene.

Transmission loss measurements* were made on a sample panel of this type of construction, having dimensions suitable for use in the Research Department transmission loss measuring facility, and predictions of sound insulation were made using the relationship given by Equation 50, Section 3.4. These predicted values are shown in Fig. 3 (solid squares). The shaded area indicates the 6 dB uncertainty in the predictions, discussed in Section 3.4. Measured values of sound insulation (open squares) are also shown in Fig. 3. These values are the mean of measurements taken in two similar vehicles (Colour Mobile Control Rooms 19 and 20) in which this type of wall construction was used. At 63 Hz and over the ranges 100–250 Hz and 500–4000 Hz the measured values of sound insulation fall within the uncertainty limits of the predictions. At 80 Hz, 315 Hz and 400 Hz the predicted value of sound insulation is lower than the measured value, while above 4 kHz higher sound insulation is predicted than was achieved in practice. In the case of the low-frequency discrepancies the shape of the "predicted" curve suggests that transmission loss measurements were affected by panel resonances which did not occur in the larger areas of wall structure used in practice. At high frequencies the practical measurements indicate the presence of leakage paths (probably door seals) which were not present (at least to

*Using Equation 35 to calculate the values from the sound pressure level differences on the source and receive side of the panel.

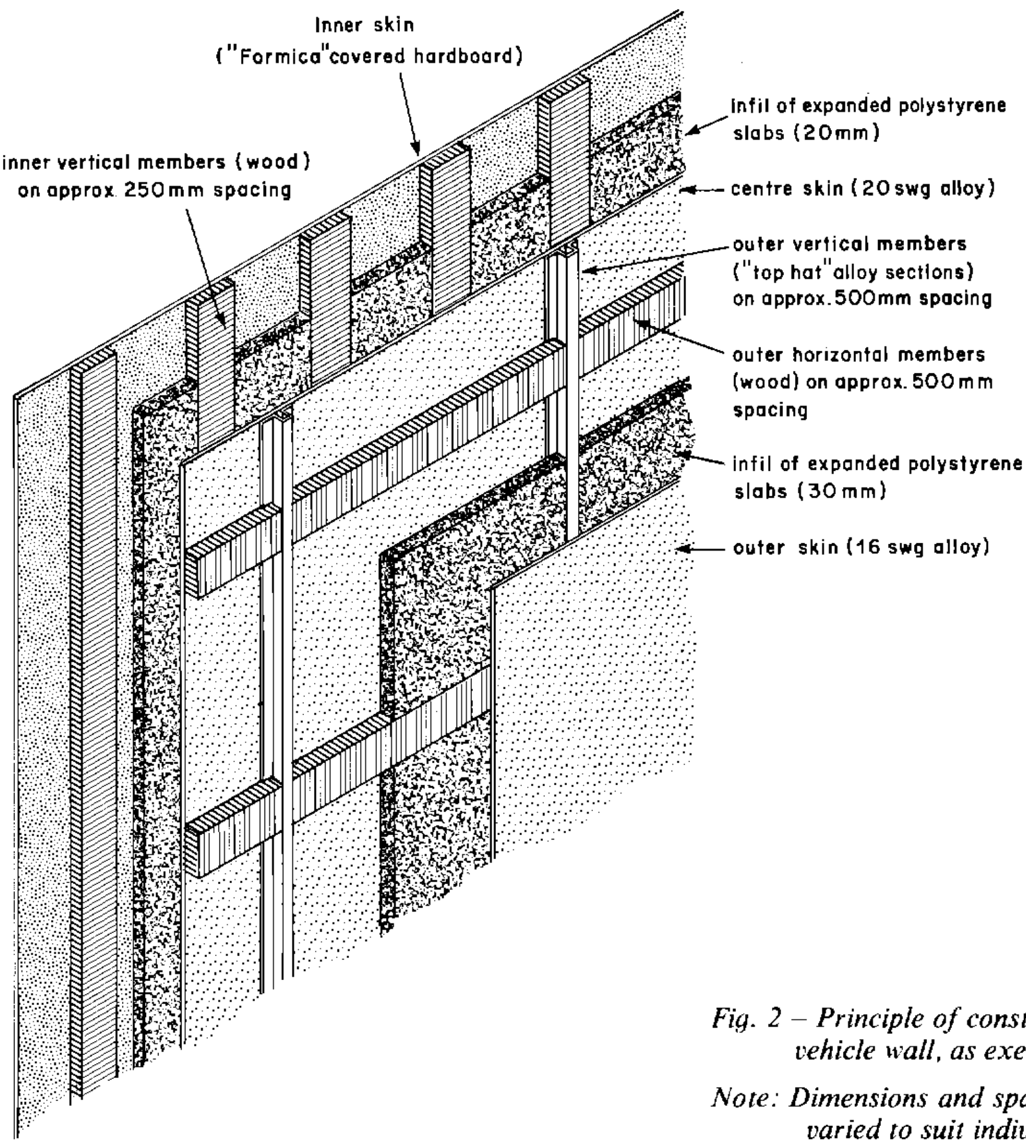

Fig. 2 – Principle of construction of present-day OB vehicle wall, as exemplified by panel 5.

Note: Dimensions and spacings of members may be varied to suit individual applications.

such a large extent) in the transmission loss measurements. In general, this comparison supports the relationship between sound pressure level difference and transmission loss, which has been suggested in Section 3.4, as applying to OB vehicle insulation and transmission loss measurements made using the Research Department facility.

5. Tests on sample panels of possible new types of Outside Broadcast vehicle wall

5.1. Details of sample panels

Assessments were made of the sound insulation that would be achieved in an OB vehicle, using four different types of new vehicle wall construction. In all cases the wall construction involved a bonding process which if necessary would enable a complete side of the vehicle to be constructed in one piece. Again, sample panels of each type of construction were made and tested in the Research Department transmission loss facility.

All four sample panels manufactured by the bonding process (Figs. 4–7) involved a "sandwich" construction, the filling material or materials being contained between outside skins which were the same in each case. The skin intended to be on the outside of the vehicle consisted of "three-ply" wood laminate 5 mm thick faced with a 2 mm layer of hard plastic (glass-reinforced plastic, or G.R.P.), while the interior skin consisted of a sheet of G.R.P. 2 mm thick. In the case of Panel 1 (Fig. 4) a lead sheet 1.7 mm thick was bonded to the interior skin, while Panel 2 (Fig. 5) was similarly constructed using aluminium sheet 0.6 mm thick. In both cases the filling of the panel consisted of rigid foam 40 mm thick, divided into strips 100 mm wide (running along the length of the panel) by thin rigid webs of

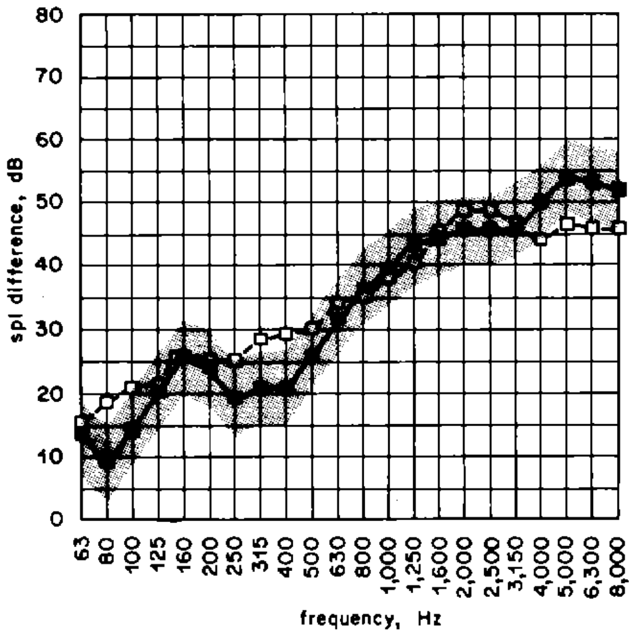

Fig. 3 – Comparison of prediction of OB vehicle sound insulation using present-day wall design (■——■) with measured insulation of vehicle with same wall design (□- - -□) Uncertainty limits of prediction.

epoxy resin. The metal skin was intended to provide electrical shielding and (particularly in the case of Panel 1) mass loading of the panel (see Section 5.3).

Metal sheets were not included in Panels 3 and 4. Panel 3 (Fig. 6) was filled with a composite layer of foam (24 mm) and rockwool (40 mm): the two layers were separated by epoxy resin 1.5 mm thick and the rockwool layer was divided into strips 50 mm wide running along the length of the panel by webs of this resin approximately 3 mm thick. The rigid foam layer was not however divided by webs in this case. Panel 4 (Fig. 7) was filled simply by rigid foam (again not divided by webs) 62 mm thick.

The surface densities of the materials used in the panels, and the effective surface densities of the complete panels, are shown in Table 1. This table also shows, for comparison, the surface density of the sample panel of present-day OB vehicle wall (referred to as "Panel 5") discussed in Section 4.

5.2. Predicted sound insulation using new types of O B vehicle wall

As in the case of the sample panel of present-day OB vehicle wall discussed in Section 4, transmission loss measurements were carried out on the sample panels of new vehicle wall construction, and predictions of sound insulation were made as described in Section 3.4. The results are shown in Figs. 8–11, which refer to Panels 1–4 respectively.

TABLE 1

Surface densities of panels under test

Panel No.	Surface densities, kg/m²				
	3-ply/ G.R.P. skin	Filling: Foam	Filling: Other	G.R.P. skin	Overall
1	6.9	1.4	13.1 (lead)	3.9	25.3
2	6.9	1.4	1.6 (aluminium)	3.9	13.8
3	6.9	0.9	9.1 (rockwool)	3.9	20.8
4	6.9	2.2	—	3.9	13.0
5	—	—	—	—	22.6

The shaded areas in these Figures again refer to the ± 6 dB uncertainty in the prediction. The measured insulation of a present-day OB vehicle is also shown in these Figures (open squares). The irregularities in the predicted characteristics that can be seen at low frequencies are most likely caused by panel resonances, as in the case of the sample panel of present-day OB vehicle wall, discussed previously (see Section 4 and Fig. 3). Here again such effects would probably not occur, or would occupy a lower frequency range, in the larger areas of wall structure that would be used in practice. The predictions that the sound insulation using the new types of OB vehicle wall would be lower than that achieved at present should therefore be treated with some reserve for frequencies below, say, 315 Hz. In every case, however, the prediction of sound insulation over a frequency range of at least an octave centred between 1500–2000 Hz is significantly worse, even taking into account the upper limit of uncertainty, than the insulation achieved in present-day vehicles. The minima in sound attenuation at moderately high frequencies which are present in the cases of Panels 3 and 4 (and to some extent in the case of Panel 2) appear typical of the type of construction used for these panels in which rigid foam is bonded to the outside skins. The rigid webs dividing this foam into vertical strips, which are present in Panels 1 and 2 but not in Panels 3 and 4, will tend to "short-circuit" the foam filling, and this probably accounts for the smaller minimum shown by Panel 2 and contributes to the presence of a "plateau" rather than a true minimum in the case of Panel 1. The presence of the rather lossy lead sheet in Panel 1 also probably contributes to the absence of a true

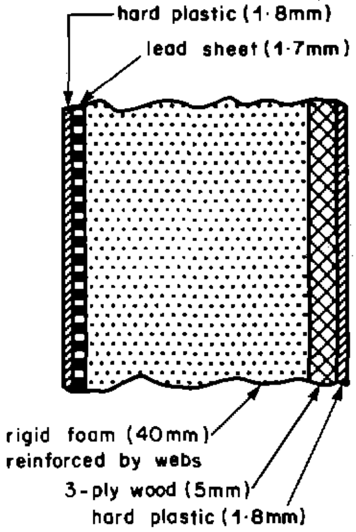

Fig. 4 – Panel 1.

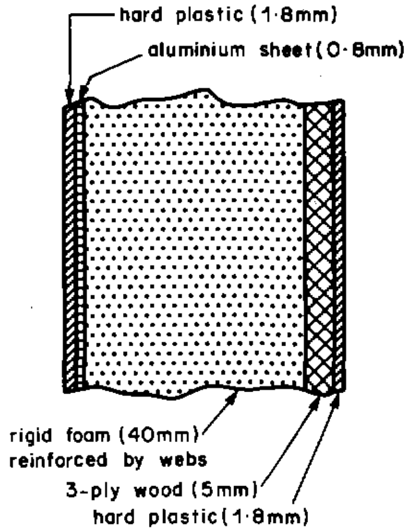

Fig. 5 – Panel 2.

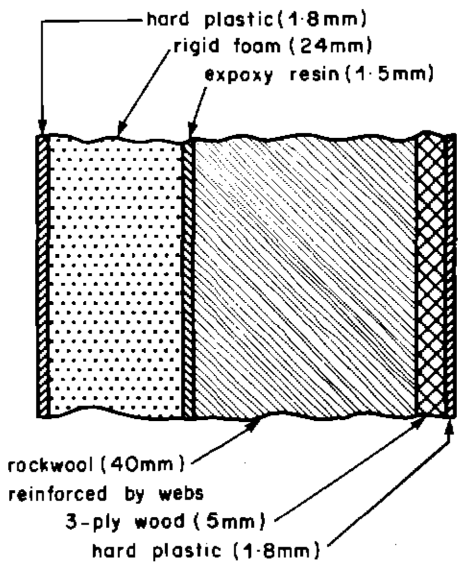

Fig. 6 – Panel 3.

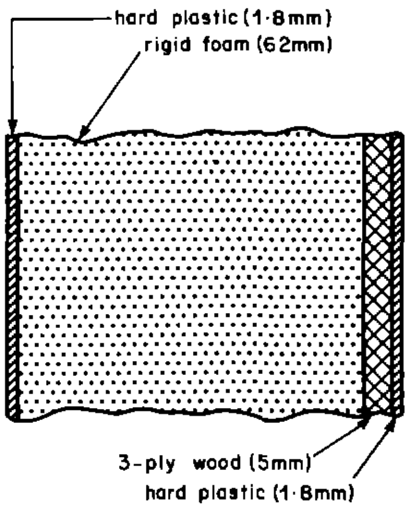

Fig. 7 – Panel 4.

Figs. 4–7 – Constructional details of test panels fabricated using the bonding process.

minimum in this case. Although at higher frequencies the Panel 3 and Panel 4 wall designs are predicted to give much higher sound insulation than achieved in practice with the present-day wall construction, such performance is of no value in the presence of the much poorer insulation discussed above. In any case, the apparent advantage shown by the Panel 3 and Panel 4 designs at high frequencies would probably be of little significance in practice, since vehicle insulation at these frequencies is determined largely by leakage paths (door seals, ventilation trunking, cable ducts et.) which inevitably occur. Since in each case the predicted sound insulation when using the new type of OB vehicle wall construction is inferior to the insulation achieved at present, it appears that this method of panel fabrication is unlikely to be suitable (at least from the point of view of sound insulation) for such purposes.

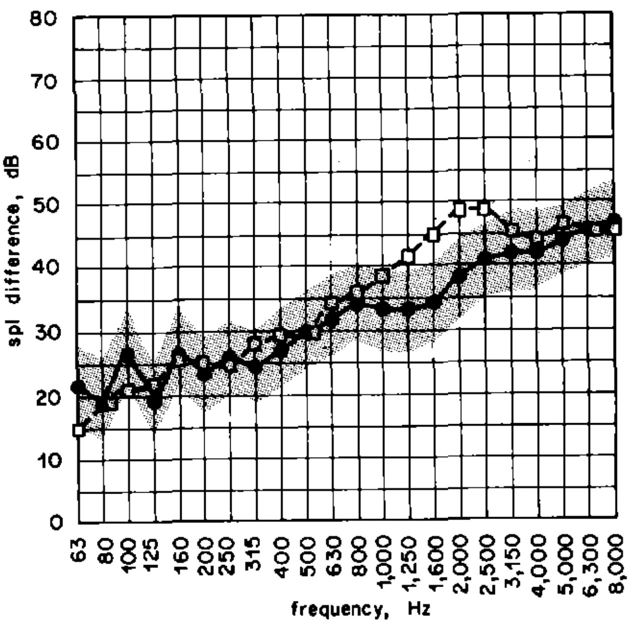

Fig. 8 – Prediction of OB vehicle sound insulation using "Panel 1" wall design (●——●), with uncertainty limits ( ). Measured insulation of OB vehicle also shown (□- - -□).

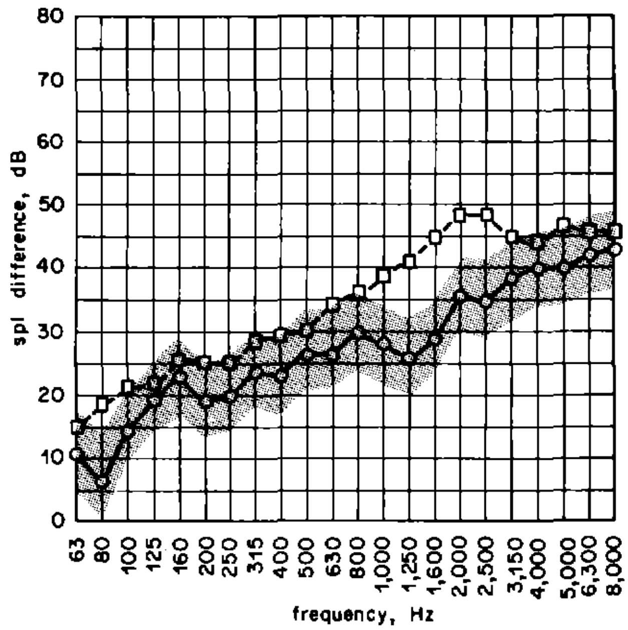

Fig. 9 – Prediction of OB vehicle sound insulation using "Panel 2" wall design (○——○), with undertainty limits ( ). Measured insulation of OB vehicle also shown (□- - -□).

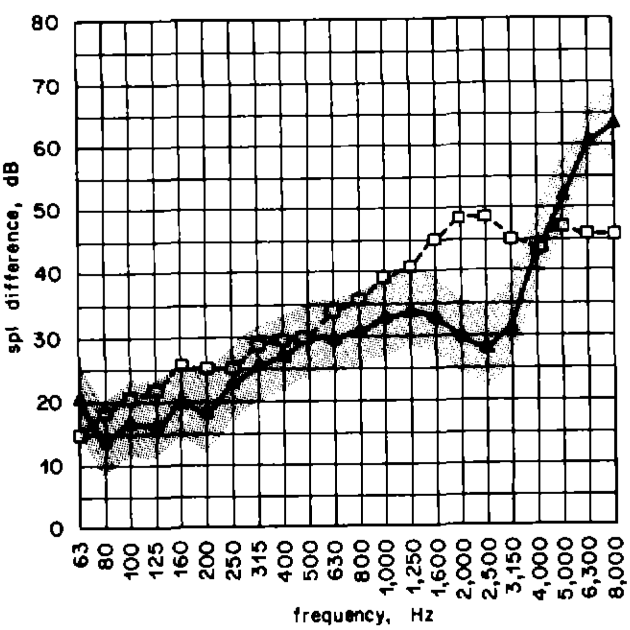

Fig. 10 – Prediction of OB vehicle sound insulation using "Panel 3" wall design (▲——▲) with uncertainty limits ( ). Measured insulation of OB vehicle also shown (□- - -□).

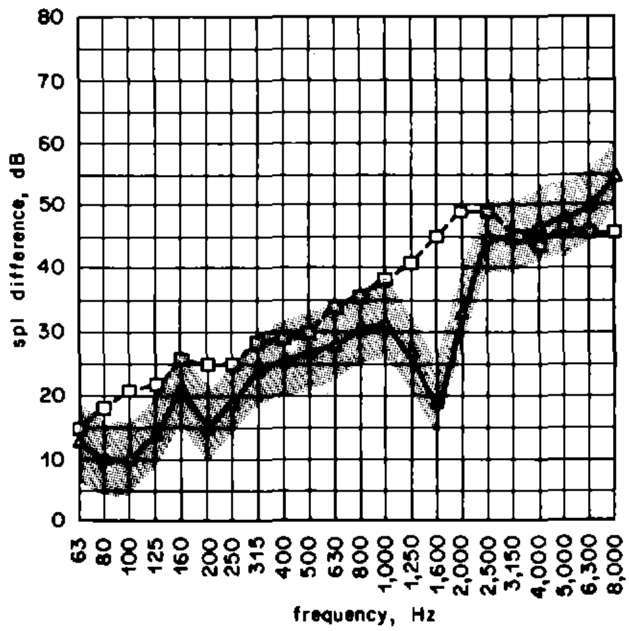

Fig. 11 – Prediction of OB vehicle sound insulation using "Panel 4" wall design (△——△), with uncertainly limits ( ). Measured insulation of OB vehicle also shown (□- - -□).

5.3. Sound insulation as a function of surface density

Provided that the specific acoustic impedance of the wall is large compared with the specific acoustic impedance of air, it can be shown[9] that in the absence of resonance effects, coincidence effects caused by the excitation of travelling bending waves, flanking paths, etc., the transmission loss of a single panel is given by

$$L_T = 20 \log_{10} \frac{\omega m}{2\rho_0 c} \qquad (51)$$

where ω is the angular frequency and m is the surface density of the panel. This relationship applies to frequencies higher than that of the fundamental panel resonance, where panel motion is mass-controlled, and Equation 51 is therefore known as the "mass law". The relationship applies to normally-incident sound radiation. It has been shown (see Section 3.3) that the sound power transmitted by a panel could be up to 6 dB greater for normal incidence than for random incidence: since Equation 35 assumes random sound incidence, the comparable version of Equation 51 therefore becomes

$$L_T = 10\log_{10}\frac{\omega m}{2\rho_0 c} + 6 \qquad (52)$$

In mks units it is usual[1f] to take $\rho_0 c = 407$ mks rayls. Thus

$$L_T = 20\log_{10}(0.00772fm) + 6 \qquad (53)$$

where the frequency (f) is in Hz and the surface density is in kg/m^2. In practice it is found that measured transmission loss is considerably lower (because of the presence of resonances, coincidence effects etc.) than the theoretical values predicted by Equation 53: this equation is useful, however, in showing that transmission loss increases (theoretically at any rate) by 6 dB for each doubling of either frequency or surface density.

Equation 53 can be written as

$$L_T = 20\log_{10}(0.00772f) + 20\log_{10} m + 6 \qquad (54)$$

and this relationship can be used to compare the extent to which the transmission losses of different panels fall short of the values predicted by Equation 53. The measured transmission loss values can be "normalized" by subtracting a quantity ($20\log_{10} m$), thus referring all values to unity surface density. (The resulting values do not in any way predict the transmission loss of panels which actually have unity surface density, since the mechanical properties of the panels themselves (e.g. resonance frequencies) are dependent upon surface density and are not modified by this normalizing process.) Fig. 12 shows the normalized transmission loss characteristics of all five measured panels, together with the normalized transmission loss predicted from the mass law relationship (Equation 54 with $m = 1$). Between 125 Hz and 800 Hz all panels show roughly the

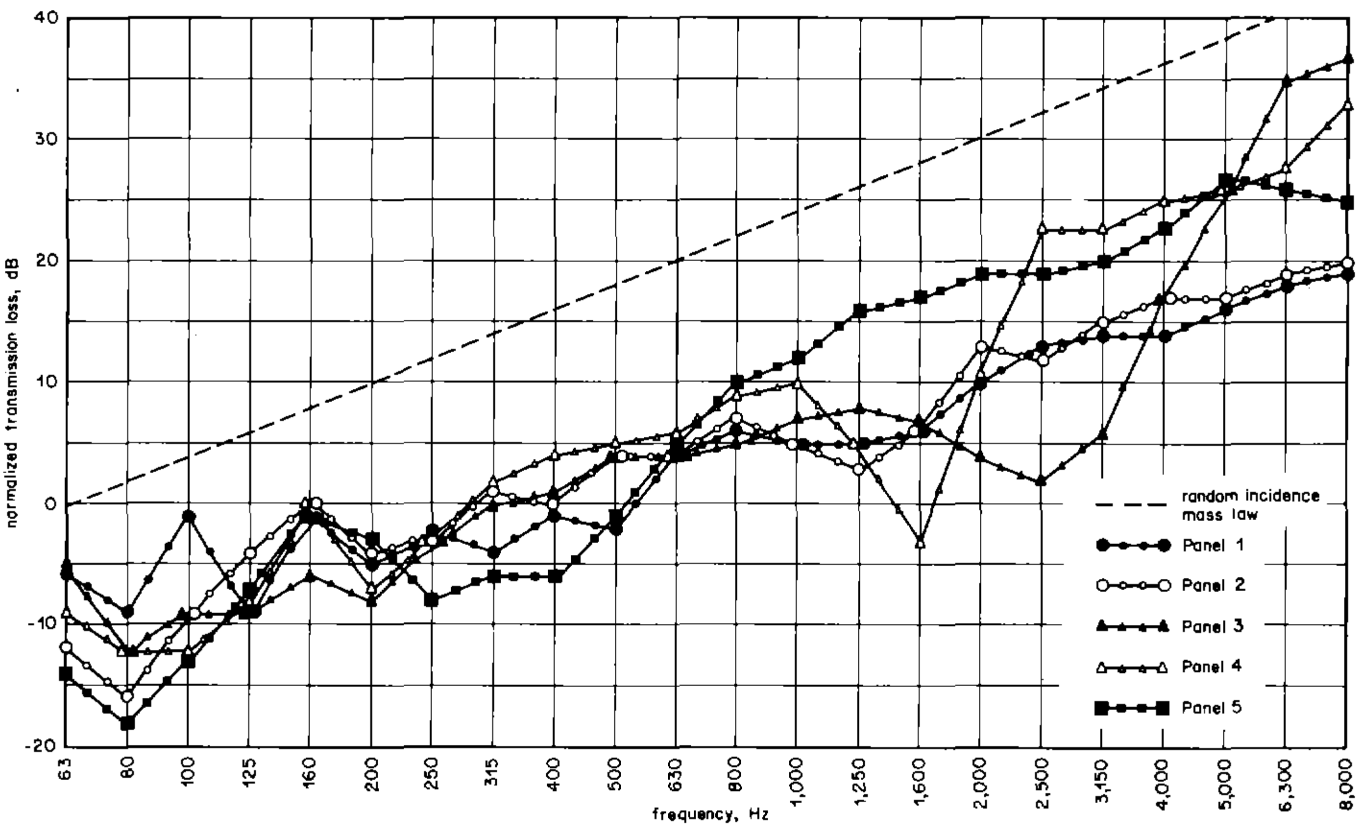

Fig. 12 – Normalized transmission loss characteristics of the five test panels.

same "inherent" performance (i.e. when due allowance has been made for their differing surface densities) which is some 15 dB below the theoretical mass law relationship. At lower frequencies Panel 1 shows better inherent performance than the others, possibly because of the rather "lossy" lead sheet built in to it. At frequencies between 800 Hz and 2500 Hz the drop in transmission loss due to the rigid foam construction (see Section 5.2) is very apparent, and Panel 5 (the sample of present-day OB vehicle wall) shows the best inherent performance. At still higher frequencies the inherent performance of Panels 3 and 4 increases markedly but Panels 1 and 2 remain inferior to Panel 5 in this respect. In fact, the very similar inherent performance of Panels 1 and 2 for all frequencies above 125 Hz is noteworthy, indicating that the differences in transmission loss shown by these two panels are almost entirely caused by their different surface densities. Overall, it appears that this conclusion applies in broad terms to all four "sandwich construction" panels, and also that this construction (or at least the materials used in it) offer no benefit (and indeed a considerable penalty at some frequencies) over and above the effects brought about by differing surface density.

The rigid foam filling used in the sandwich construction gives very little internal damping as far as transmission of sound through the panels is concerned. Indeed, a specimen of this material, excited into a bending mode of vibration, showed a "Q-factor" of about 10. The material is isotropic and this Q-factor would therefore be expected to apply to compression modes of vibration as well as to bending modes. The rockwool filling of Panel 3 might also be expected to give little internal damping; in its normal use as a sound absorbing material air is permitted to flow through its fibrous structure, but in the present case this is prevented by the impermeable skins of the panels. In addition, the rockwool is "bridged" by the rigid epoxy-resin webs which divide it into strips (see Section 5.1).

In the case of Panel 5, some internal damping is probably provided by friction between the unbonded slabs of expanded polystyrene with which the panel is filled: this may be responsible for the higher inherent transmission loss relative to Panels 1 and 2 above 630 Hz, as shown in Fig. 12.

5.4. An alternative form of wall construction

It is interesting to consider the possible construction of OB vehicle walls giving better transmission loss than is shown by the panels under consideration, while using the same bonding technique to produce wall panels which only require support at their edge. An overall surface density no greater than that of the present-day vehicle wall construction is envisaged, together with a total wall thickness not exceeding that of the thickest panel tested.

Within the foregoing constraints, the construction shown in Fig. 13 might offer better performance than any of the panels tested. A double-panel construction is shown, the panels being attached to each other and to neighbouring panels by suitably-shaped extrusions or mouldings. For greater panel rigidity the rockwool fillings could be divided by webs as in the case of Panel 3. Ideally neither inner rockwool surface should be covered, but in the interests of panel rigidity inner skins could be used, provided that they were perforated to the extent of at least 25%, after the completion of the bonding process. The two panels are made of unequal surface density so that panel resonant frequencies do not coincide (the same technique is often used in the construction of double-glazed windows), and this could be further helped by, for instance, casting the webs dividing the rockwool filling to run vertically in one panel and horizontally in the other. The gap separating the two panels is rather small compared with the spacing (at least 50 mm) usually provided in this type of construction. The two skins will thus effectively be coupled together at low frequencies to form a single panel of surface density equal to

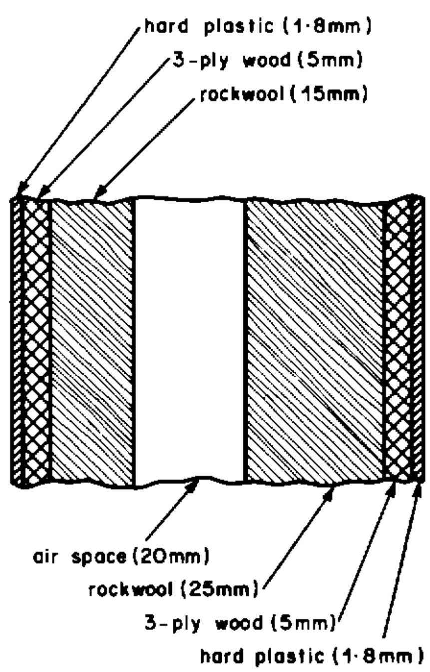

Fig. 13 – Suggested alternative design for OB vehicle wall, permitting use of bonding method of fabrication (see text).

that of present-day OB vehicle walls, but possibly having a somewhat higher transmission loss because of the wider frequency distribution of panel resonances. In addition, the transmission loss should rise more rapidly with frequency than in the case of a single panel of equal overall surface density.

6. Conclusions

The commonly-used formula for relating the transmission loss of a panel to the difference in sound-pressure level in "source" and "receive" rooms is based upon a number of assumptions:

(1) The transmitting panel occupies the whole area of one wall of the receive room.

(2) The sound field in the source room is entirely diffuse, both when making measurements of transmission loss and when using such measurements to predict sound insulation.

(3) The sound emerges from the panel in the receive room in the form of a collimated beam and strikes the opposite wall first, before subsequent reflections from other walls.

(4) Sound, after emerging from the panel in the receive room, but before striking the opposite wall (the direct sound field) occupies the whole volume of the receive room.

The first assumption is frequently not justified; the facility for transmission loss measurement at BBC Research Department, in particular, involves a specimen of area much less than the area of receive room wall into which it is set. The second assumption is probably not satisfied in the case of this Departmental measuring facility and is certainly not justified in the case of sound insulation predictions on the structure of OB vehicles. In this case, the source sound field is likely to be mainly direct, not reverberant. Although the fourth assumption is probably reasonable, it is inconsistent with the third assumption which is in any case unlikely to apply because of the presence of many modes of vibration in the transmitting panel.

Relationships between transmission loss and sound pressure level difference can be derived which do not depend on the first three of these assumptions. Uncertainties in the relationship between these two parameters remain, however, because in general the exact nature of the sound field on either side of the partition is not known. In the case of predictions of the sound insulation of OB vehicles, it seems reasonable to assume that the sound pressure level difference between the inside and the outside of a vehicle will be numerically equal to the transmission loss, but to allow a ± 6 dB uncertainty about the values obtained in this way. Predictions of OB vehicle sound insulation made from transmission loss measurements on a sample of present-day wall construction agreed with practical measurements on such vehicles within this limit of accuracy, apart from discrepancies due to known causes (particularly panel resonances and the presence of leakage paths).

Four sample panels of Outside Broadcast vehicle wall, fabricated using a bonding process, have been measured for transmission loss. Predictions of OB vehicle sound insulation made from these measurements show that the sound insulation of OB vehicles manufactured using this bonding technique will be poorer than the insulation at present achieved in such vehicles over a frequency range of at least an octave centred between 1500 and 2000 Hz. In two cases the reduced sound insulation can in part be attributed to the lower surface density of the bonded panels as compared with present-day vehicle wall construction, but in all four cases the poorer performance of these panels appears to be caused by the use of rigid plastic foam in their construction, which is believed to be responsible for dips in the transmission loss characteristics in the 1.25 kHz to 2.5 kHz range.

An alternative method of vehicle wall construction is proposed which might be expected to give better sound insulation than any of the four designs so far tested. A double-panel wall is used, each panel of which can be made using the bonding technique. Overall wall surface density and thickness have been limited to reasonable values. Other aspects of this design (e.g. thermal insulation, rigidity and cost) have not been considered.

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APPENDIX: THE SOUND FIELD IN THE SOURCE ROOM OF THE BBC RESEARCH DEPARTMENT TRANSMISSION LOSS MEASURING SUITE

The "source room" of the Research Department transmission loss measuring suite (Fig. 14) consists of a passage of length 5 m, width 1.2 m and height 3.5 mm (maximum) into which is set two flights of steps: a long flight (rise of 1.8 m) leading to access doors and a short flight (rise of about 0.6 m) leading to the test panel. The loudspeaker used to excite the source room is placed on a short landing just inside the access doors: it is thus 5 m from the panel and very approximately level with the panel's centre.

It can be shown [18] that at a distance r from a point source, the sound pressure $\bar{p}_r$ is given by

$$\bar{p}_r^{\,2} = W_X \rho_0 c \left(\frac{1}{4\pi r^2} + \frac{4}{R_S} \right) \tag{55}$$

Where W_X is the power emitted by the source, R_S the room constant of the source room (see Equation 3) and ρ_0 and c are the density of air and the velocity of sound (see text following Equation 52). The first and second terms within the brackets in Equation 55 are of interest in the present case, as they represent the contributions of the direct and reverberant sound fields respectively. Thus if $\bar{p}_{rD}$ and $\bar{p}_{rR}$ are the sound pressure levels that would have occurred if the direct and reverberant fields had been present on their own,

$$\frac{\bar{p}_{rR}}{\bar{p}_{rD}} = 4r\sqrt{\frac{\pi}{R_S}} \tag{56}$$

The value of R_S can be found from Equation 3, remembering that the average absorption coef-

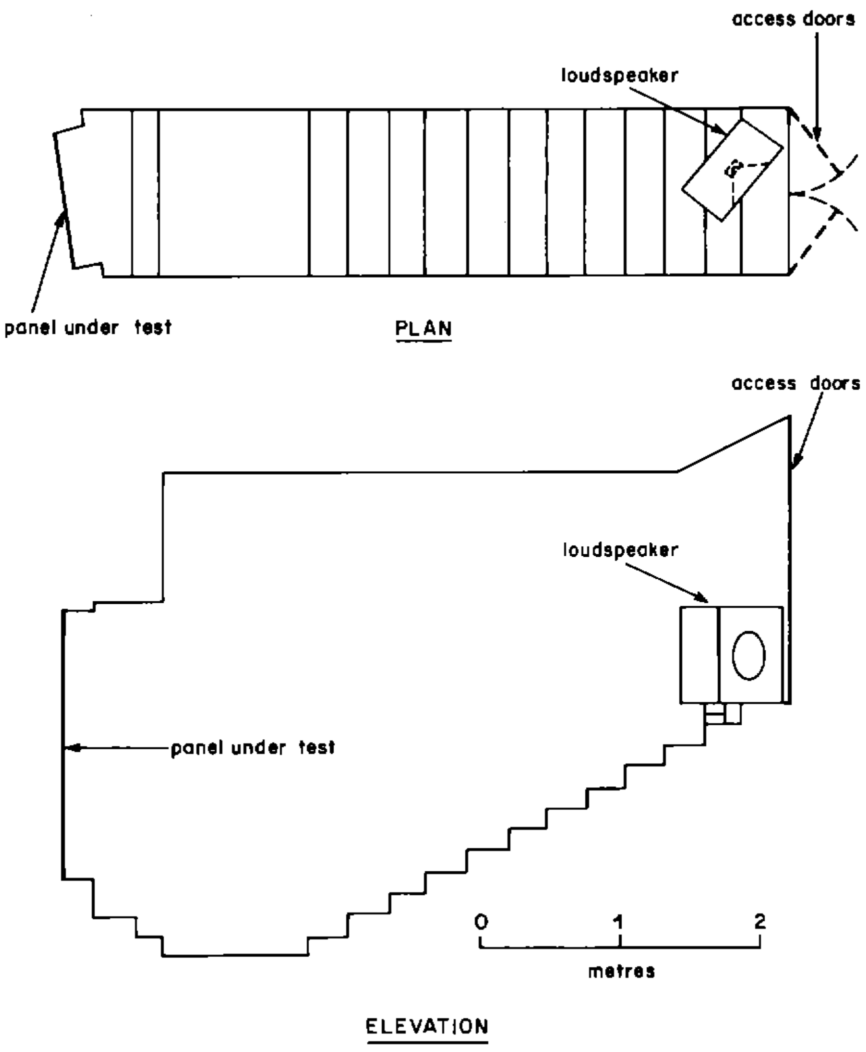

Fig. 14 – Source room of transmission loss measuring suite.

(PH-249)

ficient of the source room ($\bar{\alpha}_S$) is related to the reverberation time (T) of the room by the relationship obtained by re-arranging the Eyring formula[2b]

$$\alpha_S = 1 - \exp\left(\frac{-0.162 V_S}{S_S T}\right) \qquad (57)$$

where V_S and S_S are the volume and surface area of the source room in mks units ($16.4\,m^3$ and $46.6\,m^2$ respectively). The lowest reverberation time measured in the source room is 0.5 seconds: taking this value leads to the result in the present case that the reverberant sound pressure exceeds the direct sound pressure by a factor of 14 (or 23 dB). On these grounds the field in the source room can be taken as purely reverberant, as required in the measurement of transmission loss. The situation is however not so clear if the room geometry is considered, as the source room can be considered as a duct excited at one end and with the panel under consideration at the other. Such an arrangement would be expected to deliver plane wavefronts normal to the surface of the panel, (i.e. to set up a "direct" field) particularly when the sound wavelength was large compared with the duct dimensions. It may well be that the reverberant sound field condition is satisfied at high frequencies but not at low frequencies, although there is no evidence (e.g. an obvious "knee" or change in slope) in the transmission loss results (Figs. 3 and 8–11) to support this.

The nature of the sound field by which the panel under test is excited therefore remains ill-defined, and an "uncertainty" of 6 dB has been introduced to allow for this (see Section 3.4).

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